

AN INDEPENDENT REFINER'S APPROACH— PRACTICAL SOLUTIONS TO COMPLEX GEAR PROBLEMS

by

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ABSTRACT

Numerous failures and near failures spanning the last quarter century have occurred on a 4000 hp, single induction, double helical speed increaser in hydrogen recycle service. A persistent and unexplainable wear pattern, repeating itself consistently on the periphery of this hunting-tooth gear, eluded interpretation and correction until a program of advanced analysis techniques was applied. Modal and impact testing, lateral and torsional critical speed evaluation, as well as component redesign were utilized to mitigate the harmful affects of the complex interactions of the gear elements. The methods used to detect these interactions, and how this information was used to design a new gear will be discussed.

HISTORY

The subject gear box drives a model MGCB-333 centrifugal compressor in second stage hydrogen recycle service in the Hydrocracking Unit at the Avon Refinery, Tosco Refining Company, Martinez, California. The gear is driven by a wound rotor induction motor with liquid rheostat for variable speed control (Figure 1).

The gear was designed for an input speed of 1170 rpm with an output shaft speed of 8869 rpm. Details of the original gear construction can be found in Table 1.

Other than the initial seal failure described below, all other failures involved a severe and distinct wear pattern displayed

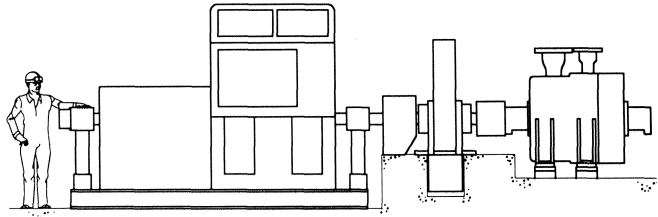


Figure 1. Motor-Gear-Compressor General Arrangement.

principally on the pinion at five tooth intervals, repeated every five teeth (Figures 2 and 3).

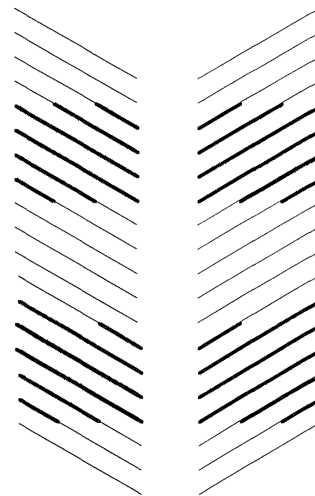


Figure 2. Cyclical Wear Pattern on Pinion Circumference.

Table 1. Original Spoked Gear Specifications.

Horsepower: 4,000	Unit Rating: 6,280 hp
AGMA Service Factor: 1.57	Total Ratio: 7.58: 1
Original Gears: 50/379 Teeth	Diameter Pitch: 5.9938
Replacement Gears: 30/227 Teeth	Gross Face: 12.5"
Center Distance: 36"	Effective Face: 10"
Universal Turbine Teeth: 30.09'04"	
H. S. Bearings: 3×4	L. S. Bearings: 7×5
Span: 22"	Span: 23.9"
Input Speed:	Output Speed:
Min. 936 rpm	Min. 7,095 rpm
Max. 1,170 rpm	Max. 8,869 rpm

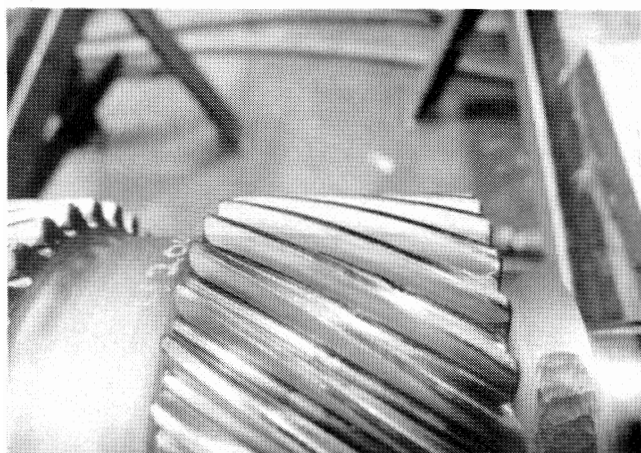


Figure 3. Photo of "Banding" Pattern on Pinion Teeth.

The following discussion on the history of the gears is provided to illustrate the pattern of failures and inconsistent methods used to detect and solve the problem.

Gear Failures

The original gear set (Figure 4) was commissioned in 1963. Satisfactory operation was maintained for over two years, wherein acoustic emissions progressively increased. In 1966, the manufacturer was asked to evaluate the operation of the unit. Sound level readings of 106 to 109 DbA were recorded. However, the company was advised that this was considered normal, and to continue operation of the unit.

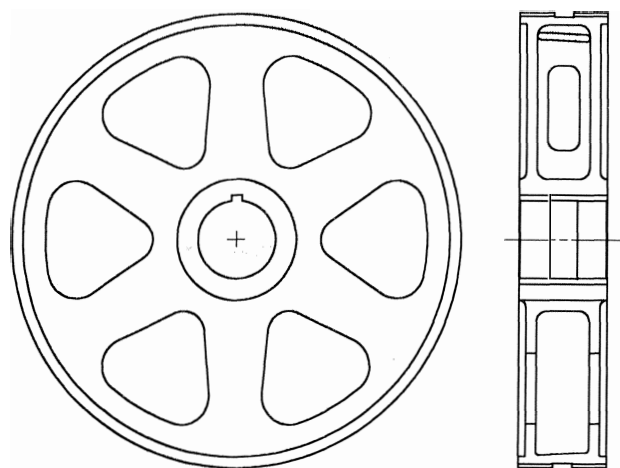


Figure 4. Original Spoked Gear.

In 1968, the gear failed catastrophically. All the teeth were shucked off one flank, and half the teeth were missing from the other. NDE revealed every fourth or fifth tooth on the gear was cracked at the root. Two spokes were cracked—one at the hub and one at center span. (The significance of these observations was not unveiled until later and will be discussed herein.)

At the time, the company had no spare for the gear available, so the gear was flown to the factory for repair. The spokes were welded, new teeth were cut and a new pinion was built. The gear was flown back and reinstalled. (This entire process was accomplished in only 11 days!)

In 1969, a replacement set of gears was received and installed. An unusual banding pattern was observed on the gear and pinion that had been in service. This is the first time the unusual wear pattern had been seen, encompassing four or five teeth and repeating itself every four or five teeth all the way around the pinion.

A year and a half later this set also became excessively noisy, and an associated cyclic beat frequency developed. An inhouse specialist performed torsional and lateral critical speed analysis. He recommended the gear be turned over and run on the opposite flank.

In 1973, the box was opened for inspection and the gear set was turned over. Inspection revealed two more cracked spokes. The unusual banding pattern noted in the previous gear set was observed on this gear as well. The original set of weld repaired gears was reinstalled and the unit returned to service.

Two years later, the gearbox had again become progressively louder, then suddenly became unbearably loud. The unit was shut down and inspected. This time, the gear spokes were cracked in three places, and there was a 0.007 in hump in the rim of the gear over the broken spokes. The banding pattern on the pinion was more pronounced and was clearly visible on the gear as well, although there were nine or ten teeth between the five tooth wear pattern, instead of the four to six tooth pattern seen on the pinion.

An order for a second replacement set of gears was placed, this time specifying courser, harder teeth, a lighter rim, and heavier spokes. The theory at this time was that the rim was trying to grow away from the spokes, causing bulges in the rim between each spoke.

In late 1976, the second set of gears was installed. At the end of two years operation, these gears also became extremely noisy and had to be changed [1]. Inspection revealed heavy wear on the pinion, two cracked spokes, a five tooth wear pattern on the gear and a six tooth wear pattern on the pinion.

Torsional Studies

Many theories were developed to explain the severe noise, cyclic beat, and heavy banding patterns on all the gears installed to date. Some thought the rim was bulging outward between each spoke. Others thought the spokes were growing out, and causing the rim to deflect.

The gear manufacturer suggested it was not their gear, but some outside influence—perhaps a torsional resonance created by the motor, coupling, or compressor.

Over the years, a total of four torsional studies were performed on the system. The first was calculated by the compressor supplier, who also supplied the gear and motor. His calculations revealed no harmful torsional frequencies in or near the operating speeds. The rotor torsional critical speed map shown in Figure 5 was supplied with the analysis results.

The second study was done inhouse, using strain gages on the input coupling. No harmful torsional resonance was discovered. However, due to the measured amplitude of the angular displacement readings, and the close proximity of the calculated first torsional to the low speed shaft rpm, the decision was made to replace the low speed coupling to move the first mode nodal point farther from the operating speed.

The third study also used strain gages applied to the input coupling. This investigation was performed by a third-party consulting firm, [2]. They concluded that there were no harmful torsional vibrations at the predominant operating frequencies. Their data compared favorably with previous studies.

The last study was performed on the entire train using telemetry. 120 tooth pulse generating gears were installed on the free end of the motor, gear, pinion, and compressor. Magnetic pick-ups measured the gate time from one tooth on the pulse

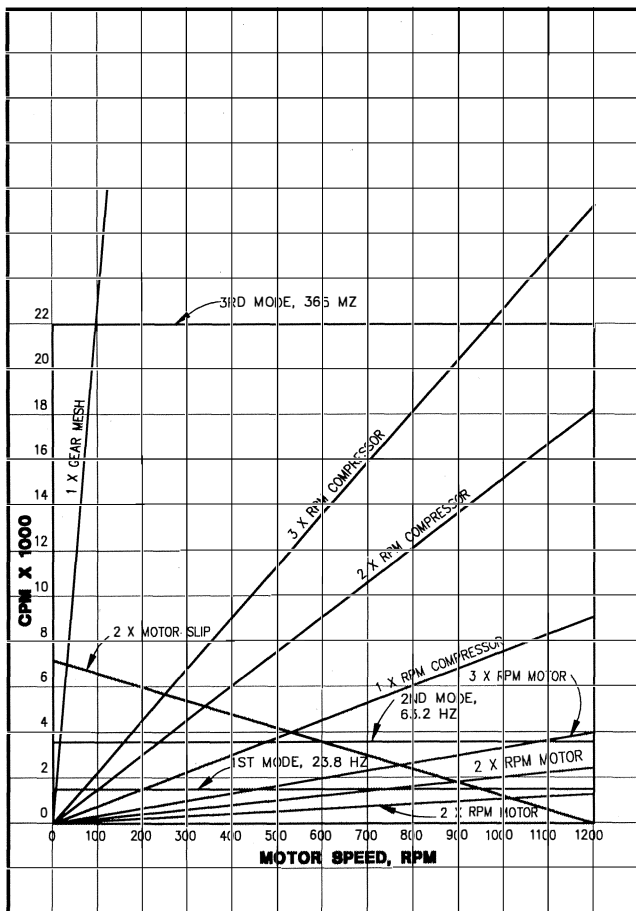


Figure 5. Rotor Torsional Critical Speed Map Including Motor, Gear and Compressor.

generating gear to the next. The signals were integrated to detect any variations, which would indicate a loading and unloading of the gears. No such variation was measured [3].

Particular attention was paid to the wound rotor motor [4], since they are known to generate a primary torsional forcing function at two times slip frequency. Two times slip is determined using the formula:

$$2 \times \text{slip freq.} = 120 \text{ Hz} \frac{(\text{Synchronous Speed} - \text{Actual Speed})}{\text{Synchronous Speed}}$$

With the 1200 rpm (20 Hz) motor, as the speed goes from 0 to 20 Hz, the $2 \times$ slip frequency would go from 120 Hz to 20 Hz. This results in a speed change factor of $6 \times$, and might explain the radical changes in gear behavior with a slight change in motor speed. However, no torsional excitation frequencies were ever measured at two times slip.

Coupling Redesigns

There were two coupling redesigns to correct possible torsional anomalies. None of them proved to have any impact on the poor operating performance of the gear.

The machine train was delivered to the jobsite with a laminated disc input coupling with a distance between shaft ends (DBSE) of $\frac{1}{2}$ in. The original torsional calculations were based on this coupling with this spacing. However, during field installation, a decision was made to provide more distance between the shafts to facilitate coupling removal. The separation was in-

creased to $5\frac{1}{2}$ in by installing a spacer piece. This effectively lowered the design stiffness of the coupling from 164,000,000 in-lb/radian to 151,900,000 in-lb/radian.

This field modification was not discovered until 1972, during a torsional study. Based on this data, the decision was made to replace this coupling with a stiffer laminated disc coupling and return to original stiffness values. The final selection had a stiffness of 244,000,000 in-lb/radian, and was calculated to cause a 10 percent increase in the first torsional resonance.

The final low speed coupling change was made in 1976, at the instance of the gear manufacturer. They were convinced the flexible element coupling could not be stiff enough to resist torsional flexing, and would also transmit axial forces to the gear.

A gear-type coupling, manufactured by the gear supplier, was strongly recommended in conjunction with the third set of gears with heavier spokes and lighter rim. This coupling had a stiffness of $1,040 \times 10^6$ and is the coupling in service today.

Bearings

The original gears had a total of three bearing modifications; two to the low speed bearings to address gear stability, and one on the high speed pinion to address a lateral critical speed resonance problem. None of these changes affected the wear pattern.

Based on the original design for torque, gear weight, and pinion reaction forces, the assumption was made that the gear would remain in the upper quadrant of the bearing under normal load conditions. Because the gears were down mesh, upward torque preload was supposed to exceed journal dead weight. In 1972, one proximity probe was installed in the axial position on the free end of the pinion. It revealed tremendous shuttling in excess of eight mils. The conclusion was drawn that the gear did not have sufficient torque loading to remain in the upper bearing quadrant. This in turn caused the gear shaft to lose the stability provided by the hydrodynamic forces generated by the oil film. This resulted in a shaft instability that created a torque induced couple motion, which caused the gear to "wobble."

To correct this problem, the OEM suggested the installation of a "Rayleigh Step," known today as a pressure dam, in the bottom half of the gear bearings. (Figure 6) It was felt this would provide the additional upward force required to keep the shaft in the top half of the bearing, as per the original design assumption.

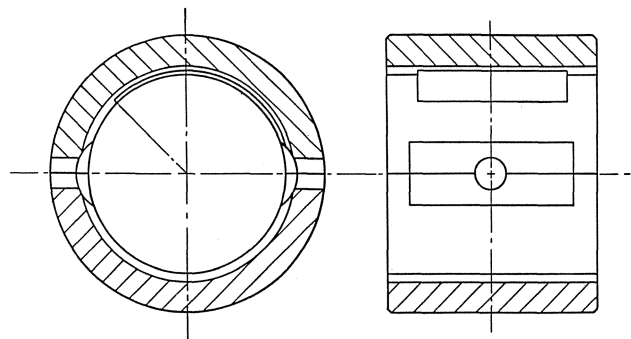


Figure 6. Pressure Dam Bearing Used In Lower Half of L.S. Bearing.

In 1978, the entire gear was instrumented with proximity probes and the first comprehensive vibration analysis survey was performed using state of the art real time analyzers. This study concluded that the gear was in a conical whirl mode caused by inadequate bearing support.

Through discussions with process engineers, the company discovered the purity of the recycle gas had improved, lowering its specific gravity. This reduced the amount of torque required to pump the gas, which caused the gear to drop as gravity force exceeded the uplifting effect of the torque.

The recommendation was to put the pressure dam in the upper half of the gear bearing to work with gravity to hold the bearing down. The new pressure dam bearings were installed in 1979. This change had some stabilizing effect, but the noise and vibration levels continued to propagate.

In 1980, a new reformer unit was commissioned. This caused the purity of the recycle hydrogen to improve again, unloading the gear further, increasing its instability. The pinion shuttling mode increased, as it tried to follow the wobbling path of the gear face. The severe axial impact created by this process excited the second critical speed of the pinion.

The final bearing change was performed on the pinion, in an attempt to move the critical out of the excitation range. Using finite element analysis, a pinion model was created (Figure 7). The model was then used to determine the optimum stiffness required to move the critical mode outside the operating range. A five pad tilt-pad bearing appeared to have the best chance for stabilizing the pinion. The bearings were installed and tested but analysis of the test data revealed a reduction in the critical speed of merely six percent, which did not stiffen the pinion enough to move the critical outside the operating speed.

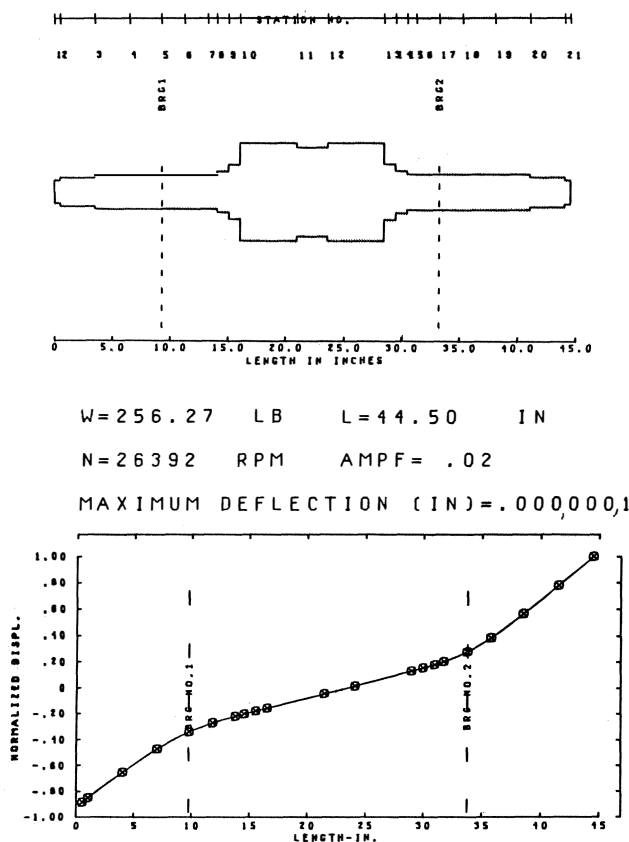


Figure 7. Pinion Model and Unbalance Response.

REDESIGN GEARS—FIRST ATTEMPT

By 1981, the gears were being operated at reduced speed in an attempt to keep the pinion from shuttling. At the same time,

gas purities were rising, putting pressure on operations to increase compressor speed. In addition, the gear OEM insisted the problem was external and not a problem with their design which, by the way, they didn't use anymore!

Losses in production were calculated (\$6K/day) and the edict was issued to stop trying to make the existing gears work, and instead buy a gear so radically different that it could not possibly exhibit the same behavior as the old gear. Little did we know.

The decision was made to use a solid disc design for the gear, with enough mass to keep it solidly in the lower quadrant of its bearings. A vendor was selected based on the ability to supply such a gear and design it to fit the existing footprint. They were also willing to cooperate with the company, which was very important in the light of previous experience with the supplier of the original gear.

A consulting firm was hired to do the torsional and lateral natural frequency and bearing optimization studies [4]. Three design cases were studied:

- *Fully Hogged Gear*—This gear would be heavily profiled, with a thin web.
- *Partially Hogged*—This gear would have a moderate profile, with a substantially thicker web.
- *Unhogged Gear*—Essentially a solid disc, with no profiling in the web area.

Lateral critical speed maps were generated for the gear shaft and pinion (Figure 8). The study concluded that as long as bearing stiffnesses were maintained above 250,000 lb/in., excitation of the gear shaft should not be expected.

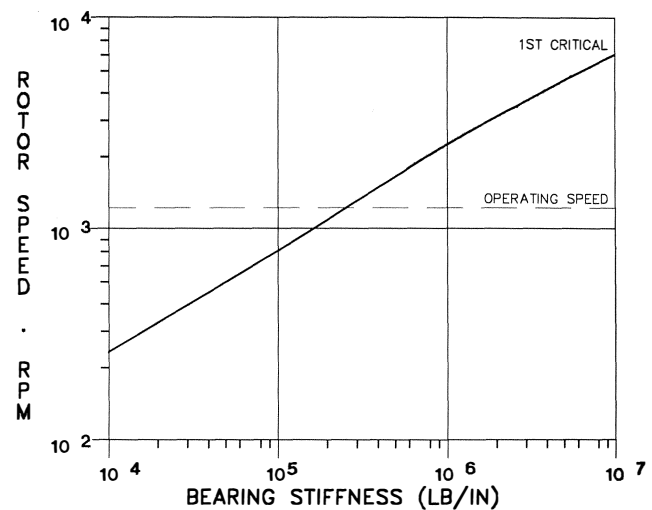


Figure 8. Lateral Critical Speed Map.

The results of the bearing load analysis are shown in Figures 9 and 10. For each bearing, the total load is the vector sum of the static weight of the rotor and the gear reactions. Gear reactions include the transmitted load, which is tangential to the pitch circle, the radial load, or separating force which is normal to the pitch circle, and an axial component which is equal to zero in this case since the gears are double helical. The old gear is shown in Figure 10, with the load very close to the split line. This is an unstable region due to the presence of the oil distribution groove, and could contribute to instability. The gear chosen to replace the existing unit is shown in Figure 10. Notice at 100 percent load the resultant vector (W) remains in the lower quadrant.

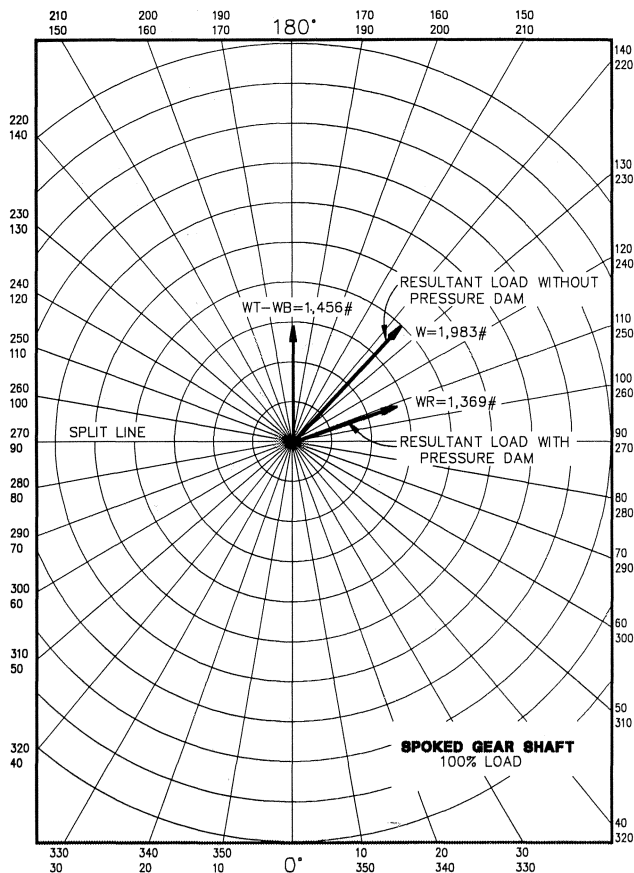


Figure 9. Bearing Load Vector—Original Spoked Gear.

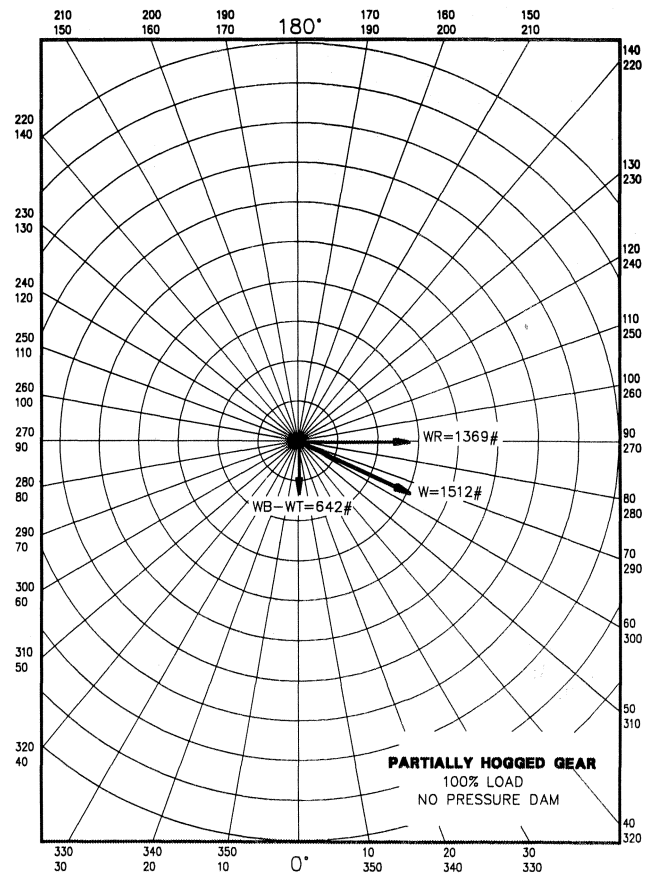


Figure 10. Bearing Load Vector—Partially Hogged Gear

PARTIALLY HOGGED GEAR

Based on the preliminary analysis performed, the decision was made to use the partially hogged gear design (Figure 11). It had the greatest separation from the torsional critical while exhibiting good bearing stability at 100 percent load (Table 2).

Table 2. Torsional Analysis Results.

Case	Gears	L. S. Coupling Stiffness (in/lbs) rad	Torsional Natural Frequency (cpm)	Case
1	Spoked	152×10^6	1208	Original Coupling
2	Spoked	244×10^6	1310	Thomas Coupling
3	Spoked	1020×10^6	1427	Present Coupling
4	Hogged Bullgear	1020×10^6	1532	
5	Hogged Bullgear	1020×10^6	1346	
6	Partially Hogged Bullgear	1020×10^6	1410	

In 1982, an order was placed on a fast track to meet the next turnaround outage. An additional gear blank was ordered at the same time in case casting flaws were discovered in one of the forgings.

The new gear was installed in 1983. Less than one year later, the company notified the manufacturer that the gear noise had risen dramatically since startup to 111 DbA.

In late 1984, the plant was taken down briefly for catalyst skimming. While down, the inspection cover on the gear case was lifted. To the utter amazement and dismay of the users, the

exact banding pattern that we had observed on all the spoked gears had appeared on the pinion of the new solid disc design.

Immediately after the gear went back into service a thorough vibration survey was done. The results showed a predominate gear vibration frequency at 1570 Hz. (Figure 12) The vibration consultant concluded that there were definitely symptoms of increasing gear resonance at that frequency.

The gear design consultant was asked to confirm the company's field measurements [5]. A finite element model of the gear was developed. The first umbrella mode was reported at 1576 Hz., confirming the actual field test data.

During the next scheduled maintenance outage, the gear and pinion were removed. Several of the upper case half holddown bolts adjacent to the gear bearings were found broken completely through from fatigue stress, an indication of the energy being dissipated through the bearings.

The pinion bearings were severely damaged. This damage was initially thought to be electrostatic discharging, but was later identified as cavitation. (Figure 13) The gear bearings showed incipient signs of cavitation as well.

The pinion was flown back to the factory for examination. The tooth profile was measured across the width and breadth. Wear of 0.0015 in was recorded in both planes. These are hardened gears that have only been in service for 22 months!

Modal Analysis

The firm that does the vibration analysis recommended a set of structural frequency response tests be performed in order to determine the frequencies, damping, and mode shapes associated with the gear system. This type of testing is generally

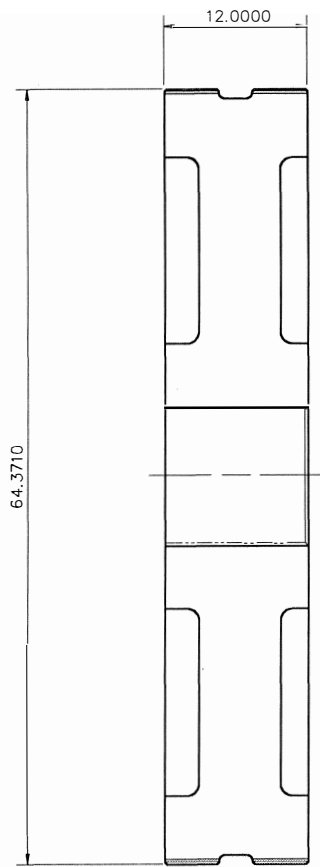


Figure 11. Partially Hogged Gear Profile.

referred to as “modal analysis” or “impact testing,” wherein the gear is excited via a calibrated impact hammer. The force applied to the system during impact, along with as the response of the structure are recorded during testing [6].

The ratio of the Fourier transforms of the system’s response to the impact results in an engineering unit of response, acceleration (Gs) in this test, per pound of force at each indicated frequency from the spectra generated by the impact. Further, processing of these spectrum via a structural dynamics analyzer produced the dynamic mode shapes displayed herein.

Testing was performed on the partially-hogged (profiled) in service gear, the blank (unhogged) forging ordered at the same time the profiled gear was purchased, and the old spoked gear previously in service.

The specific objectives of the testing were: (a) to establish all prominent normal modes of the gear structure; (b) compare the previous spoked gear dynamic characteristics to the new gear design; and (c) to determine by testing the spare blank, the extent of hogging and profiling, if any, required to render a gear with structural characteristics less likely to promote the ob-

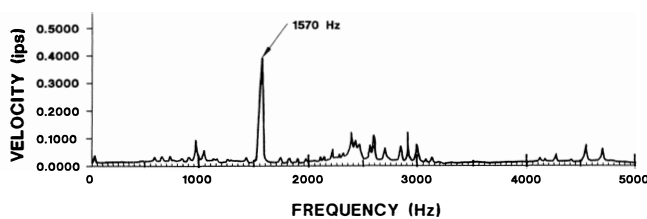


Figure 12. Spectra Showing 1570 Hz Gear Resonance.

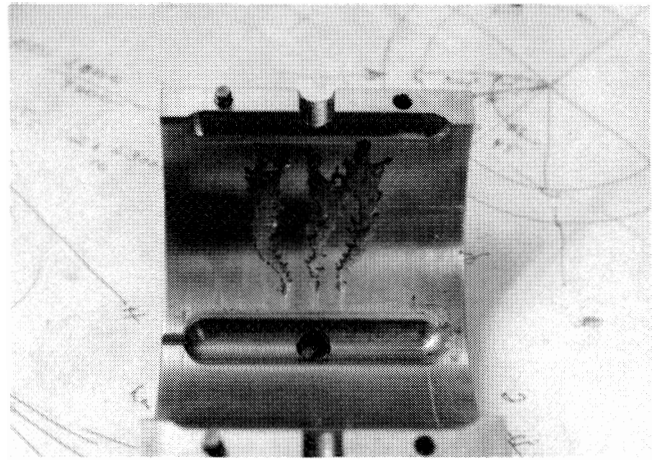


Figure 13. Cavitation Damage to High Speed Bearings.

served resonant frequencies, that could prompt the unrelenting wear patterns of all previous gears.

Test Results

Despite the magnitude of dynamic signal acquisition, computer generated analytical work, and transfer function testing, no absolutely conclusive and exact mechanism for the observed failure history and wear pattern generation has been found.

Common gear structural characteristics and observed pinion behavior strongly suggest a highly complex interaction between the two gears. Specifically, this involves excitation of a pinion critical at approximately three times operating speed which may be coupled to a nearby cymbal mode resonance in the gear.

Excessive pinion axial motion and gear “wobble” has been observed and calculations allow the possibility that the wear pattern may be closely phased resonances between 480 to 530 Hz. The actual frequency response (FRF) to the applied force is shown in Figure 14, indicating that a resonance condition is present in the gear [7].

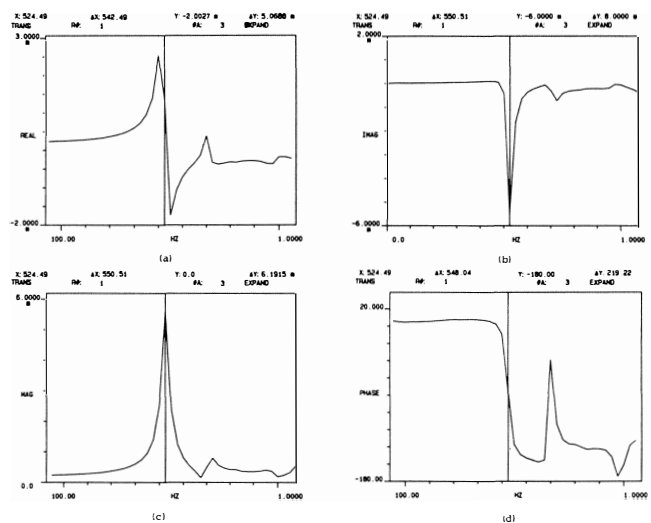


Figure 14. (a) Real Component of Displacement, Indicating the Amount of Response IN PHASE; (b) Imaginary Component of Displacement, Indicating the Amount of Response OUT OF PHASE; (c) Log. Magnitude of the Forcing Function; (d) Phase Lag of Displacement Behind Force.

The principle modes observed were all of the "cymbal" type, ranging from the simple umbrella modes to multiphased edge scallops, and span a frequency range from approximately 500 Hz to 4,100 Hz, excluding the lower frequency rigid body modes of the gear assembly [8]. See Table 3 and discussion under UNHOGGED GEAR.

In summary, the study identified three principle modes, of which the lowest 524 Hz mixed mode of umbrella and edge scalloping appears to be the most closely related to the observed wear pattern (Figure 15). This component is close to a sometimes prominent operating response at 480 Hz, seen in the gear data under advanced wear conditions, and is also close to a critical speed of the pinion as determined by earlier analysis.

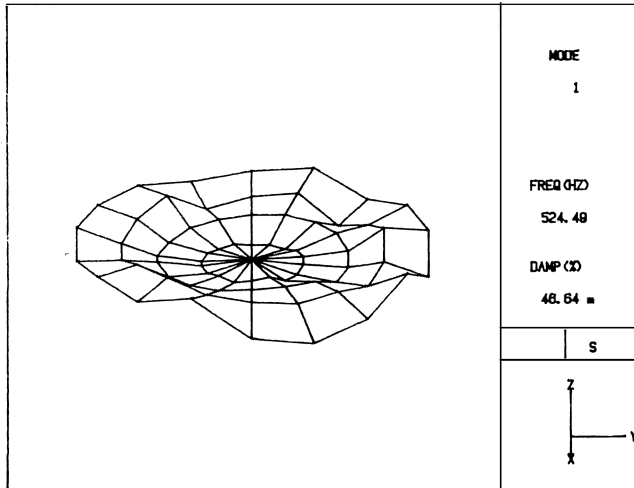


Figure 15. Scallop Mode Shape at 524 Hz From Partially Hogged Gear Modal Analysis. Pinion second critical resides very close to this frequency domain. Note non-synchronous nature of the scallop shape at the gear periphery.

The composite modes exhibited in the profiled gear at 524 Hz were those of an umbrella mode coupled with approximately 3.6 scallops around the periphery and could be synchronized with the excitation force. It is the interplay between this function and the pinion critical (at or about three times operating speed) which represents the most likely wear mechanism.

This hypothesis was reinforced using a computer program developed for prediction of hunting tooth type calculations. The calculations were performed in an attempt to discover possible reoccurring high energy tooth reactions based on various combinations of interplay between the edge scalloping effects and pinion motion. As would be expected, the results showed excessive sensitivity to frequencies selected, but clearly showed that such combinations were possible at the approximate frequencies observed.

The impact testing also indicated principle gear responses at 1,750 Hz and 4,110 Hz. (Figures 16 and 17) The former frequency was identified as the first umbrella mode during the design review by the gear consultant, but is definitely a second order umbrella mode. The degree of scalloping at the edge is minimal, and the frequency is not consistent with any known mechanism which could produce the "banding" type wear pattern.

The high frequency 4,110 Hz mode was of interest because of its proximity to the actual gear mesh frequency of 4,300 Hz. However, this mode was found to be another higher umbrella mode of the 3rd or 4th order, and did not correlate with the generation of the wear pattern. This mode is lightly damped, which

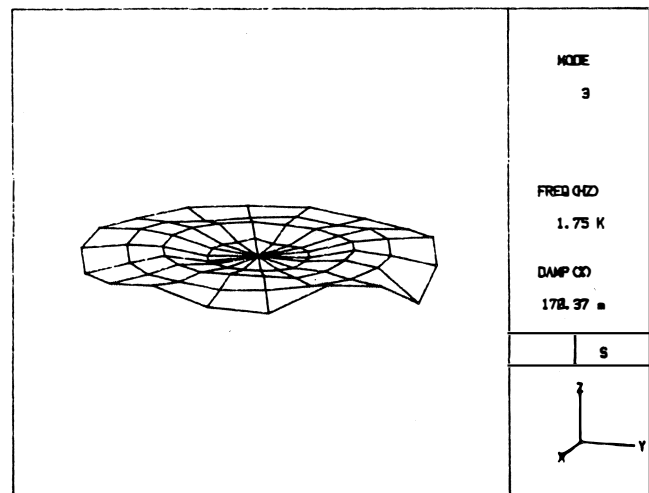


Figure 16. Initially Identified as the First Umbrella Mode, Impact testing revealed this response at 1,750 Hz to be a second order umbrella mode, not attributable to a banding pattern.

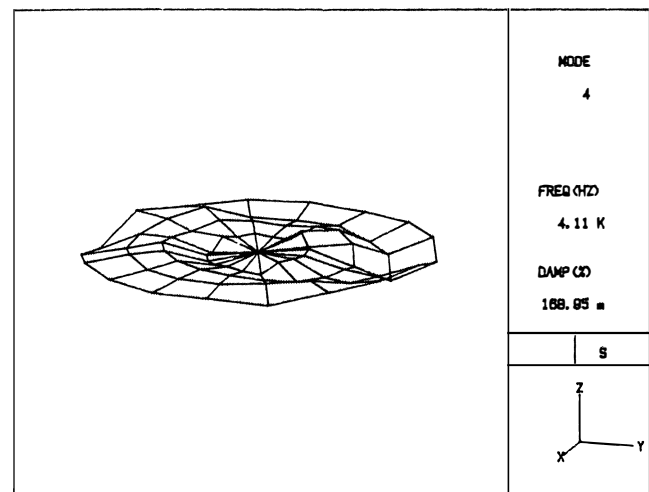


Figure 17. 4,110 Hz Mode is Close to Gear Mesh Frequency. Testing found it to be a third or fourth order umbrella mode. Its light damping could accelerate wear, but was not responsible for the banding pattern.

could explain its emergence when surface wear on the teeth advances.

SECOND REDESIGN—UNHOGGED GEAR

The results of impact testing clearly corroborated the fact that the partially hogged gear was not suitable for reliable longterm operation. The testing also indicated that the blank forging may provide a viable alternative solution, at least in the interim, until another gear design was developed.

A comparison is shown in Table 3 between the partially hogged and unhogged gear, with respect to the frequency of principle mode shapes and the percentage of critical damping measured at those frequencies.

Some interesting results emerged from the impact testing. First, the scallop mode shape at 524 Hz on the partially hogged gear is nonexistent in the unhogged disc. Instead, the unhogged disc at 530 Hz exhibits a much more symmetrical mode shape

comprising two sine wave patterns at the periphery of the disc, with no perceptable axial (umbrella) type motion observed (Figure 18). Axial motion, described as cymbal or umbrella motion, is greatly reduced on the unhogged gear at all mode frequencies.

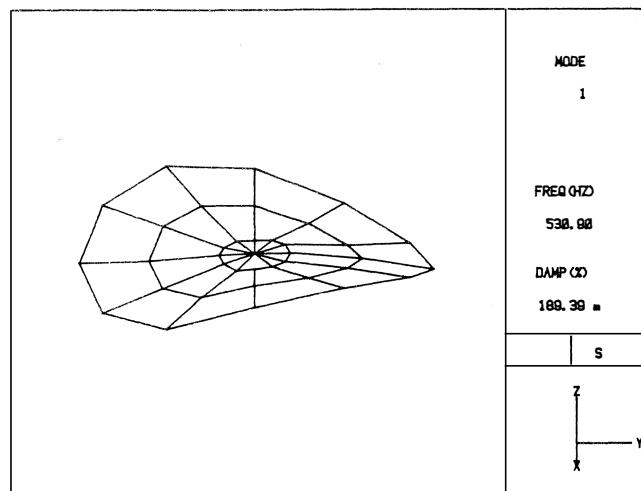


Figure 18. The Unhogged Disc at 530 Hz Exhibits a Much More Symmetrical Shape, Characteristic of a True Umbrella Mode.

Also, it appears the mode shapes have shifted frequencies on the unhogged disc. The mode shape of the unhogged gear at 530 Hz more closely resembles the mode shape of the partially hogged gear at 1,094 Hz and so on as the second, third, and fourth modes manifest themselves.

A comparison is shown in Table 3 between the partially hogged and unhogged gear, with respect to the frequency of principle mode shapes, and the percentage of critical damping measured at those frequencies. The percent of critical damping, which indicates the rate at which an impact is dissipated in a structure, is substantially higher in the unhogged gear.

The residues from the damping values at the rim, web area, and hub are graphed in Figure 19 for the partially hogged and unhogged gears, clearly indicating the ability of the unhogged gear to operate more stably in the presence of an excitation force.

The impact test data results were fed into a signal processing computer. Based on the location of the sensor and the magnitude of the response to the impact, the computer is able to generate a dynamic model that displays the actual motion of the gear at various vibratory frequencies.

The computer display was videotaped and presented to the gear manufacturer. Reaction to the problem was as unpredictable

Table 3. Comparison of Partially Hogged and Unhogged Gear Cymbal Modes and Damping.

Cymbal Mode	Partially Hogged Gear		Unhogged Gear	
	Freq (Hz)	% Damping	Freq (Hz)	% Damping
Mixed Scallop	524.491	46.635m	—	—
1st Order	1.094	—270.240m	530.898	189.392m
2nd Order	1.745K	170.365m	1.117K	—276.214
3rd Order	4.112K	168.955m	1.736K	587.400
4th Order			4.345	361.367
1st Order				

Note that although frequencies are similar, the unhogged gear has "shifted," displaying more conventional mode shapes. The mixed scallop mode of the partially hogged gear seems to have "displaced" the 1st cymbal mode.

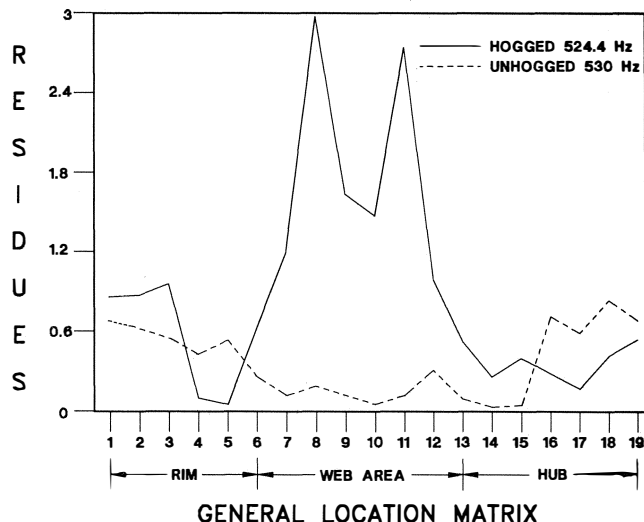


Figure 19. Comparison of the Partially Hogged and Unhogged Gear Transfer Function Residues at Their Respective 524 Hz and 530 Hz Excitation Responses. Points 1-6 were measured on the rim, points 7-12 in the web area, and points 13-19 on the hub.

ble as the behavior of the gears. They agreed that the analysis had some merit, and would be willing to work with the user company to help solve the problem!

They put their inhouse dynamicist to work on modelling the gear using a finite element analysis program. They found bending modes at 474 Hz and 497 Hz. Damped response analysis was also performed using the bearing stiffness and damping coefficients at operating speed. The first bending mode frequency occurred at 525 Hz, precisely what was measured in the field. Their work is shown in Figure 20 [9].

Specification

Based on the positive indications from the test and analytical data, the decision was made to manufacture a gear using the

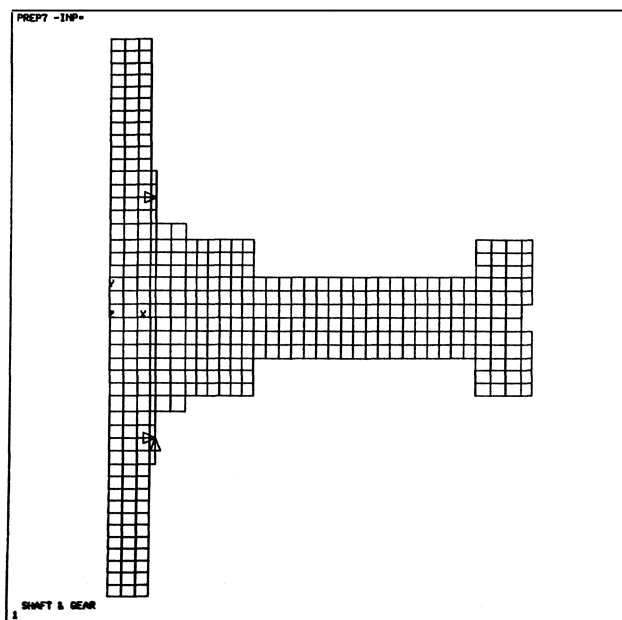


Figure 20. Finite Element Analysis Performed by the OEM.

blank disc on hand. Prior to placing the order, all of the work that had been done by the gear consultants for the partially hogged gear selection was reviewed. Several additional requirements were specified prior to issuing the purchase order.

- Pinion and gear were to be hardened (carborized and nitrided respectively) and ground to equivalent American Gear Manufacturer's Association (AGMA) quality 13 [10]. Nitriding enhances wear resistance thus ensuring long term conjugate gear action, minimizing dynamic loading and vibration.

- The face (between tooth flanks) and side of the rim were to be prepared as probe target areas per API 670, for verification of unbalance and wobble amplitudes [11].

- The axial stability check, also known as apex runout, shall be held to 0.001 in TIR, API Standard 613, Special Purpose Gears for Refinery Services specifies 0.0015 in [12].

- Thin film, trimetal bearings with four tapered lobes will be used to provide additional strength and resistance to cavitation. Bearing design will be reviewed to optimize stability [13].

During the data review, two more benefits were realized by using the heavier, unhogged gear. First, the gear shaft would remain firmly in the lower quadrant of its bearings under all load conditions (Figure 21). Second, the additional mass of the gear changed the stiffness of the shaft, and moved the shaft critical speed out of the operating region.

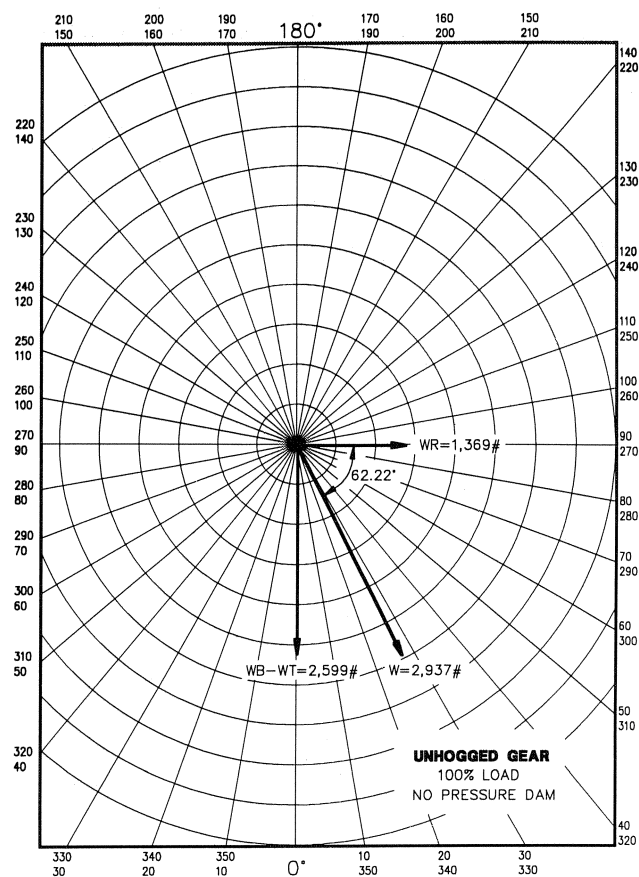


Figure 21. Bearing Load Vector—Unhogged Gear.

Testing

The blank gear was sent to the factory and the gear teeth machined, ground and honed. The tooth-to-tooth accuracy, pitchline runout, lead/profile graphs, accumulated error charts,

mechanical/electrical runout and face contact tape liftings were all performed to acceptable industry and company standards.

However, because the gear weighs over 11,000 lb, it was difficult to support it rigidly enough to get consistent apex runout readings on the pinion. To get the support stiffness required, special bearing supports were designed, cast, and machined to adapt to the existing gear checker (Figure 22). The test was then successfully conducted to the user company's specification of 0.001 in TIR.

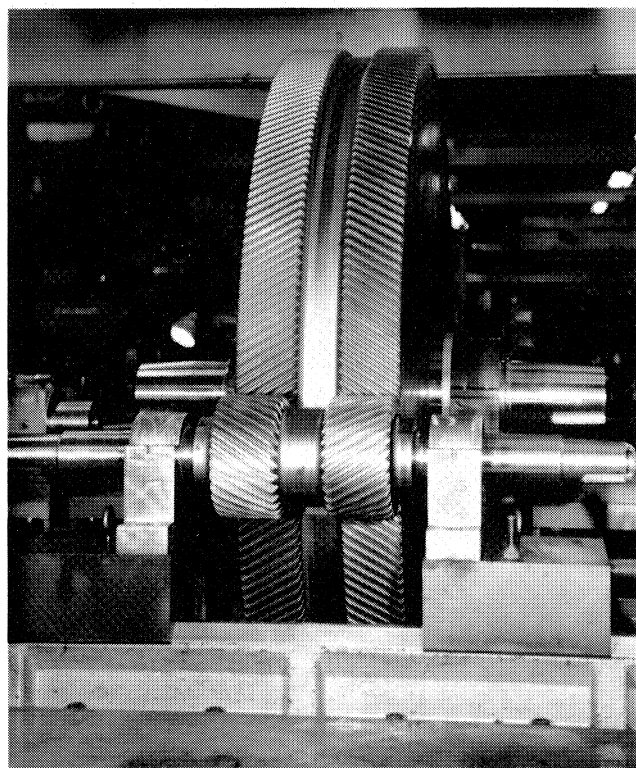


Figure 22. Unhogged Gear in Custom-Built OEM Device for Testing Apex Runout. Six ton gear was within specified tolerance of 0.001 in TIR.

High Speed Coupling

Two predominant schools of thought exist on the precise mechanism causing the wear on the pinion. One theory says the wear is caused by the pinion shuttling axially, as it tries to follow the wobble motion and scallop mode shape of the gear. Another claims it is the complex interplay of the pinion lateral critical and gear mixed resonance mode. In the author's opinion, they are both true to some degree. In either case, the pinion has to follow the path of the gear. Coupling design could have an impact on pinion's ability to track the gear.

The original high speed (output) coupling was a continuously lubricated, gear-type design. The American Petroleum Institute has stated that "Several coupling types . . . have the potential for transmitting high axial forces as thermal or load changes cause connected shafts to grow toward each other or try to separate. Only in the gear-tooth and grid types is this axial force indeterminate . . ." [14].

Under full load conditions, one could theorize that the pinion and compressor rotor could be in effect one mass element. The pinion has to overcome the inertia of its own mass as it shuttles to keep up with gear gyration. It must also drag the mass of the

compressor rotor (with its aerodynamic axial forces) along with it!

To reduce the effect of the situation described above, the decision was made to install a flexible element coupling. Flexible element couplings are extremely compliant axially. Several designs were studied: Single element membrane, multielement membrane, and multiple (laminated) disc.

Stiffness was a major concern in the final design, to avoid any possibility of torsional resonance. Another torsional critical speed analysis was performed to determine the effect of the new coupling [15]. The study revealed the existing coupling yielded a torsional of 10,077 rpm, uncomfortably close to operating speed. The users were reluctant to increase coupling stiffness, however, since that would have a tendency to reduce damping and create a more sensitive system.

The decision was made to put in a softer coupling that would provide a first torsional critical below minimum operating speed. By doing this, the users could measure it, watch it, and put it well outside operating speed. 7,000 rpm was selected as the desired torsional. Based on a coupling weight of 40 lb and a WR^2 of 200 lb-in², the required torsional stiffness would be 2.1×10^6 lb-in/radian.

The multiple disc coupling appeared to have the greatest potential for modification to meet the desired stiffness, comparable weight and overhung moment. Disc couplings are easier to maintain, inspect, and service. The supplier also has the capability to verify the stiffness by test. The desired stiffness was specified in the purchase order. A standard design hub and element were used, with a "tuned" spacer to achieve the proper stiffness value (Figure 23).

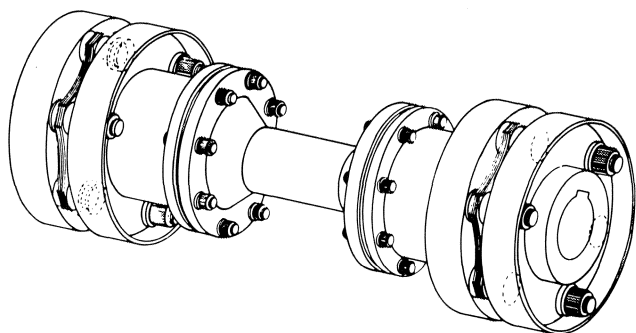


Figure 23. Flexible Element Coupling With "Tuned" Spacer, Designed to Yield a Specified System Torsional Stiffness.

It was critical that this value be correct, so the user specified that the coupling stiffness be tested and witnessed. The test was performed by attaching one side of the coupling to earth, and torquing the other end while measuring the force applied and amount of deflection (Figure 24). After correcting for the final coupling weight, the resultant test stiffness yielded the desired system torsional speed.

UNHOGGED GEAR INSTALLATION AND OPERATION

The unhogged gear was installed during a planned maintenance turnaround in February 1989. To assure a successful installation, optical alignment checks were made prior to shutting the machine down, and afterward when the machines were cold. The alignment of the gear case bearing housings relative to each other were also checked optically. After the compressor was reinstalled, prior to connecting the piping, the compressor was laser aligned, using the offset values measured during cool

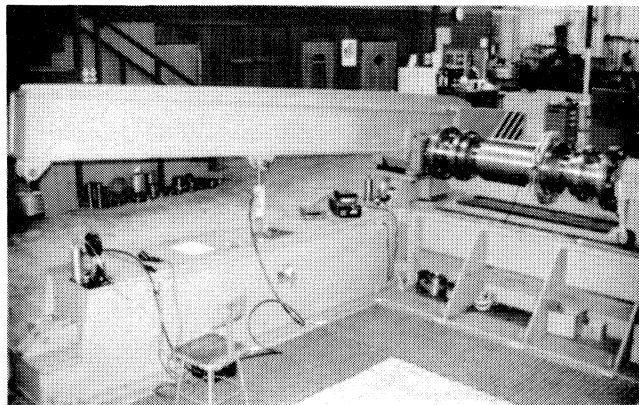


Figure 24. Device used to Apply Torque and Measure Stiffness for the H.S. Flexible Element Coupling.

down, and during the manufacturer's recommended running position calculations for the gears in the gear casing.

The new "tuned" coupling was installed with the compressor rotor held against its active thrust bearing, the gear in its assumed running position, and the pinion firmly meshed into the gear teeth. No prestretch was used, since the measured temperature rise was not enough to cause significant axial growth. All three axial thrust probes were zeroed.

Additional vibration probes were installed in the gear case coincident with the target areas on the side of the rim and center of the gear face, to measure axial and radial movement during operation.

To improve wear resistance, the trimetal bearings were installed, and the lube oil was changed from turbine oil with an ISO rating of 32 centistokes, to an ISO 46 AW (antiwear package) turbine oil.

The gear was recommissioned in March 1989. At full speed and load, they were much noisier than expected. The users noticed that the gear axial probe indicated a shift of 20 mils from its prestart position. The unit was shut down and the coupling position was readjusted axially to compensate for the observed movement. When the unit gear was returned to service, the noise had completely subsided.

Based on the axial stiffness data supplied with the coupling, 20 mils of displacement equates to about 30 lb of force. The users were surprised that amount of force could have such a dramatic effect on the dynamics of the system. This experience demonstrated the potential the original gear coupling had to create a similar yet more profound effect on pinion behavior.

A complete set of vibration signatures was acquired in March 1989, after the unit reached normal operating condition. The results indicate the gearbox performance with the new elements is excellent, with good synchronous and gear mesh responses. Pinion stability appeared good and the gear wobble observed in the past was much abated. The resonance response primarily responsible for poor gear performance in the past was undetectable and it is assumed that the radical design changes have successfully eliminated this problem. This belief was further reinforced when the annual followup analysis, conducted in March 1990, indicated little or no deterioration based on comparison with previous data. The absence of the resonant frequency response is shown between the hogged and unhogged gear in the vibration spectra (Figure 25).

CONCLUSION

The advent of sophisticated methods and machines for testing and measuring complex signals has opened the door to solving

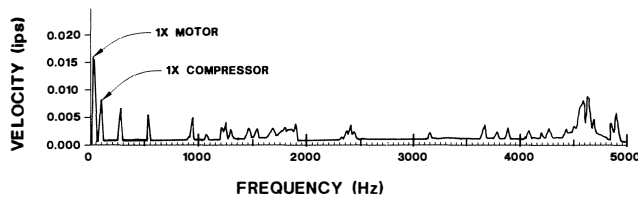


Figure 25. Most Recent Vibration Spectra From the Unhogged Gear Show the Absence of Harmful Resonant Frequencies.

extremely perplexing machinery problems like the one discussed here.

The combination of empirical data and computer simulation gives an engineer the opportunity to hone a model until its behavior parallels that of the actual machine. The users discovered that using dynamic signals from a running machine and comparing them to a model were not enough. They were not able to identify and define troublesome gear resonance and poor damping qualities until they embarked on a systematic program of modal analysis.

The final elements in a formula for success are patience and perseverance—yours and upper management's! A lot of effort was put in to finding out what the problem was not. Eliminating all the possibilities can become a time consuming, expensive process. However, if the problem is approached in a systematic way, using all the tools available to the analyst today, the odds for successfully solving a complex problem such as ours are in your favor.

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